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HEAT-DRIVEN FAN FOR TENT HABITABILITY IMPROVEMENT

BY
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This exploratory development effort investigates one method of improving cold weather tent habitability, the circulation of heated air. Tests were carried out to determine the quantity of air necessary to substantially reduce the thermal gradient in a GP Medium and Frame-Type Expandable tent. It was found that 5-W was enough power to operate a 30-cm diameter fan at 700 RPM, which reduced the thermal gradient 62%. A 5-W thermoelectric module was purchased, and finned heat exchangers were designed to provide a 200°C temperature difference across the module, using the space heater exhaust as a heat source. Although an

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20. ABSTRACT (cont'd)

appropriate low voltage motor could not be located, a prototype demonstration model using a poorly-matched motor achieved 80% of the desired fan speec, proving the feasibility of a heat-driven fan.

PREFACE

A means of improving cold weather habitability of tents is presented. Other studies have investigated passive means of improving habitability such as the addition of liners or insulated floors. This study investigates the use of forced air circulation to increase comfort by reducing the substantial thermal gradient. It is an effective way to make the best use of the available heat with little penalty in terms of setup time, weight, and bulk.

Appreciation is expressed to Mr. John Roche for his help in the experimental portion of this study.

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HEAT-DRIVEN FAN FOR TENT HABITABILITY IMPROVEMENT

INTRODUCTION

In cold weather, a tent can be quite an uncomfortable place. The military field space heater, non-electrically powered, can only transfer heat by radiation and natural convection. A column of heated air rises briskly above the heater to the uppermost part of the tent. The natural buoyancy of the heated air sets up a substantial vertical thermal gradient. In a heated General Purpose (GP) Medium Tent for example, temperatures on a –10°C (14°F) day are typically 28.3°C (83°F) at the peak, 21°C (70°F) at head level and 7.5°C (45°F) at lower leg level. The uninsulated walls and earth floor are considerably colder than the air temperature in the tent, and the radiant heat loss of the inhabitant to these surfaces also adds to the discomfort. The most difficult part of the body to keep warm, the lower legs and feet, are in the coldest part of the tent. The habitability of the tent is marginal even though the average tent temperature is in the confortable range. This study investigates the use of air circulation by a fan to reduce the thermal gradient, and the use of exhaust gas heat to generate thermoelectric power to operate the fan.

This study began as an Independent Laboratory In-House Research (ILIR) effort in which a Stirling engine was first considered as the means to drive a fan. The study transitioned to an exploratory development project in November 1980. The Stirling engine concept lost out to thermoelectrics when it was found that only about 5-W of power was needed. Also contributing to this decision were possible problems with Stirling engine vibration and noise.

RECENT STUDIES

Improved tent habitability was the primary goal of two recent NLABS studies. Pilsworth studied one of the most tangible factors relating to habitability, heat loss.[1] He developed a method to calculate heat loss to reduce the amount of time-consuming experimental work involved in evaluating alternative tent designs. For a GP Medium Tent, he calculated typical heat losses to be: walls 42%, roof 24%, floor 17%, and air infiltration 17%. He discussed the reduction of heat loss due to addition of a tent liner, a wooden platform floor, insulated walls, and a tarpaulin floor in quantitative terms, but discussed comfort only in qualitative terms.

Barca experimentally investigated the effects of various liners and line configurations on the thermal microclimate of a frame-type tent.[2] His report also includes a good discussion of previous studies, referencing research dating back to 1943. In the winter portion of the

[1] M. Pilsworth, Jr., The Calculation of Heat Loss from Tents, US Army Natick Research and Development Laboratories, Natick, MA, Technical Report NATICK/TR–79/017, September 1978 (AD A072415).

[2] F. Barca, Thermal Comfort in Tents, US Army Natick Research and Development Laboratories, Natick, MA, Technical Memorandum, July 1981.

studies, two tents were erected in an outdoor location and were instrumented to measure thermal performance. One tent was used as a control, and the other was modified with various liners. Liner materials were: cotton oxford cloth, aluminized plastic film, and 2.5-cm thick fiberglass with cotton cloth facings. Floor liner materials were a 1.3-cm thick urethane foam with neoprene facings (used with fiberglass liner) and a single-ply vinyl. Liners were attached to the inside walls, ceiling, floor and peak in several combinations. Data indicated the cotton oxford liner on the walls and ceiling, with a vinyl floor, reduced heat loss 61%; the insulated wall, ceiling and floor liners reduced heat loss 78%. Barca concluded the major causes of heat loss were air infiltration and exchange, and a high convective heat transfer coefficient. The addition of a simple cloth liner provided a semi-dead air space to reduce the convective heat transfer coefficient and a second fabric barrier to reduce air infiltration.

Comfort was increased by having the tent inner surface temperatures closer to the inside air temperature so that radiant loss from the occupant was reduced. The insulated liners were much more effective in increasing comfort in this manner. The greatly reduced heat loss provided by liners contributed to increased comfort by permitting air temperatures throughout the tent to be considerably higher. The thermal gradient was not appreciably reduced by any of the liners in any configuration, except for the one case with insulated wall and floor liners. Here the liner created an open box within the tent so that passage of cold air over the top of the box tended to cool the air at 1.8 m, and thereby reduce the 0.3 to 1.8 m gradient.

The use of tent liners does increase the habitability and does reduce heat loss. For a given tent temperature comfort is slightly increased with cloth or reflective liners, and is increased considerably with an insulated liner. The increase in comfort level of an insulation lined tent, however, comes with a decided increase in erection time, added expense, and extra weight and bulk. The weight and volume of the insulated floor, ceiling and wall liners for one 4.9 x 4.9 m tent is 75 kg and 1.33 m^3. In addition the fiberglass must be kept dry to maintain its effectiveness.

In the aforementioned studies, the thermal gradient was measured and discussed, but was never effectively reduced. The objective of the present work was to determine the effectiveness of air circulation in reducing the thermal gradient, and to design a means to accomplish the circulation under field conditions in tents heated with Army non-powered heaters.

REDUCTION OF THERMAL GRADIENT WITH FANS

Use of a fan to circulate the heated air from a space heater is not a new idea in the Military. The Barracks Heater, a 20.5-kW (70,000-Btu/hr) oil-fired military space heater, has an optional 25-cm-diameter electric fan that can be mounted on the ceiling above the heater. This fan operates at 1500 RPM and is rated at 18.4 m^3/min volume flow rate. It is powered by a 1/20-HP (37-W) electric motor from a 115 V AC source. Electricity is unavailable in tents in the field, but there is an abundance of heat from the space heater to provide thermoelectric power. The first task was to conduct an experiment to determine the amount of air circulation necessary to substantially reduce the thermal gradient.

The first heat stratification test was carried out in a frame-type, expandable, 4.9 by 4.9-m tent. An M1941 liquid fuel space heater was positioned under the stack vent hole in the roof. A 30-cm diameter, three-bladed, 30° pitch angle air-circulating fan was mounted to the frame cross-section directly over the heater, with airflow directed downward. This fan was the Patton "High Velocity" air circulator, model G-1272, with low, medium, and high speed positions. The low speed setting was used, and a Seco variable transformer was used to adjust the fan speed to the desired value; fan speed was measured with a Strobotac strobe light. A wooden thermocouple pole, with unshielded copper-constantan thermocouples mounted 0.3, 1.2, and 1.8 m above the ground, was positioned 2-m away from the heater. A separate thermocouple was mounted at the tent peak. Figure 1 shows the test setup.

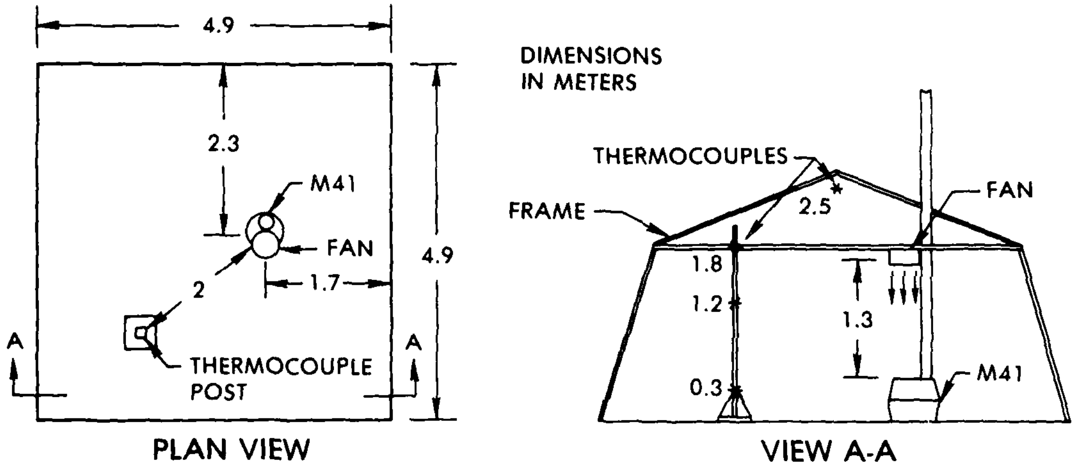


Figure 1. Test Setup to Measure Thermal Gradient in Frame Type, 4.9 x 4.9 m Tent

The low fan setting resulted in a rotational speed of 990 RPM; this speed created an airflow that seemed excessive, causing too much draftiness. Fan speeds of 600, 700 and 800 RPM were tested for effectiveness in reducing the thermal gradient. Temperatures were recorded on a Honeywell Elektronik 112 temperature recorder, with range −25 to 75°C.

The test procedure was to light the heater, turn on the temperature recorder, and let the tent warm up for at least an hour. The heater firing rate was adjusted so the tent temperature would be in the comfortable range. The fan was then turned on at the lowest test speed, 600 RPM, and allowed to operate for approximately 1/2 hour. The fan speed was then increased to 700 RPM for 1/2 hour, and the 800 RPM for 1/2 hour. Temperatures were recorded continously throughout the test. Data for this test are located in the Appendix, Table A–1. Original data for all tests covered in this report were recorded in NLABS Laboratory Notebook 7478. A condensed summary of Table A–1 test data is shown here in Table 1.

Table 1. Thermal Gradient v.s. Fan Speed, Frame Type, 4.9 x 4.9 m Tent

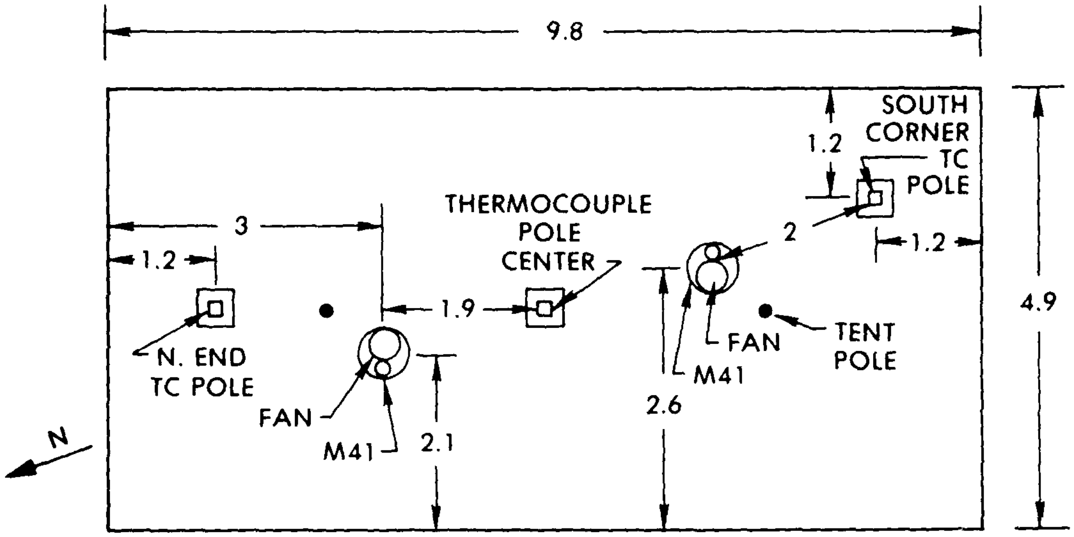
Fan Speed (RPM)	Ambient Temp (°C)	Thermal Gradient (°C/m) 0.3–1.8 m	0.3–2.5 m
0	4.0	5.7	5.7
600	5.2	2.7	1.2
700	5.5	2.3	0.9
800	5.7	2.3	0.7

The thermal gradient of 5.7°C/m at zero fan speed is considerably less than the value of 9.1°C/m measured by Barca (reference 2). This is probably due to the relatively warm ambient temperature and the moderate rainfall that occurred during the test. As a point of reference, according to Barca (reference 2), a thermal gradient of approximately 1.8°C/m is considerable acceptable in a typical frame-type home. Considering the tent has an earth floor and thin uninsulated walls, as a first order approximation a thermal gradient double this value, or 3.6°C/m, would be a realistic goal.

As the zone of occupancy of a tent is between the ground and about 1.8 m (6 ft), the thermal gradient referred to in discussions throughout this report will be over the interval 0.3–1.8 m above ground level.

The fan speed of 600 RPM reduced the gradient 53%, but higher fan speeds reduced it only slightly more, 60%. The gradient at all three fan speeds was below the 3.6°C/m goal discussed above. The 0.3–2.5 m gradient was much lower than the 0.3–1.8 m gradient due to the fact that the 2.5-m temperatures were one to two degrees C lower than the 1.8-m temperatures. This inverse gradient is probably due to the air currents set up by the fan, and the moderate rainfall affecting the temperature at the tent peak. A decision was made to run this test in a larger tent.

A General Purpose Medium Tent, measuring 4.9 by 9.8 by 3.0 m high, with 1.7 m eaves, was erected. This tent has an internal volume of 150 m^3, more than three times the 46 m^3 volume of the frame type tent. Two heaters were required to heat the tent. A plan view of the instrumented tent is shown in Figure 2.



DIMENSIONS IN METERS

Figure 2. Test Setup to Measure Thermal Gradient, GP Medium Tent

Three wooden thermocouple poles were set up: one in the center [illegible]ding upwards 2.9 m with the top about 10 cm from the center beam, one in the no[illegible]d, and one in the south corner, both extending upward 2.2 m with tops about 6 cm from the roof fabric. Unshielded copper-constantan thermocouples on the center pole were at the 0.3, 1.2, 1.8, 2.5 and 2.9-m heights, while the other two poles had thermocouples at 0.3, 1.2, 1.8 and 2.2-m heights. The ambient air thermocouple was on the east side of the tent, 2 m away and 1.2 m above the ground. The tent walls were sealed at ground level by placing 5-kg sandbags on the ground flap approximately 1 m apart. There was a building on the east side of the tent, about 15 m away. Wind speed and direction were measured with an Air-flo Instrument windspeed and direction transmitter, Model T–420A, the same instrument used in the reference 2 study. The unit was located about 30 m to the NW of the tent.

Two M1941 space heaters were used, fired with gasoline, and two 30-cm-diameter Patton fans were attached to the heater stacks 1.2 m above the top of the heater. One fan was fitted with a DC permanent magnet motor, a Torque Systems Model MH–2130–001A. This motor was previously used as a servo motor on another project; it is rated at 93 W continuous output, which is many orders of magnitude more power capability than is needed for this test. It was used to get some idea of the DC power draw necessary to operate the 30-cm diameter fan. The other fan used the AC motor that came with the fan, and a Seco Variable transformer was used to vary the speed. A Lambda DC power supply, Model LK341A was used to power the DC motor, a Data Precision Model 245 multi-meter was used to measure amperage, and a Universal AVO voltmeter was used to measure voltage. A Strobotac strobe was used to measure fan speed, and temperatures were recorded on the Honeywell Elektronik 112 temperature recorder. An Omega Trendicator temperature sensor, Model 410A–T, was used to measure inner-stack and stack surface temperatures 1 m above the heaters, on the side opposite the fan. Iron-constantan thermocouples were used. This information was used to balance the output of the two heaters.

Three preliminary tests were run to check out the instrumentation and to determine which fan RPM values to use. Three formal tests were then run; the data for these tests are in the Appendix in Tables A–2, A–3 and A–4. The three fan speeds chosen were 600, 700 and 800 RPM. The data contained in Tables A–2, A–3 and A–4 were averaged, and a summary is included here in Table 2.

Table 2. Average Thermal Gradient v.s. Fan Speed, G.P. Medium Tent

Fan Speed (RPM)	Avg. Ambient Temp (°C)	Avg. Thermal Gradient (°C/m)		Gradient Reduction (%)	
		0.3–1.8 m	0.3–2.9 m	0.3–1.8 m	0.3–2.9 m
0	–7.2	8.9	7.7	–	–
600	–4.2	4.4	3.4	51	56
700	–8.5	3.4	2.4	62	69
800	–8.2	2.9	1.7	68	78

A 51 to 68% reduction of thermal gradient is indicated for the three fan speeds. Examination of Tables A–2, A–3 and A–4 data in the Appendix reveals a quite uniform reduction in thermal gradient at all three measurement locations. The increase in comfort was very noticeable. On the coldest test day, –13°C (Table A–4), the fans operating at 700 RPM increased the near-ground (0.3 m) temperature 8.5 to 11.5°C above the fan-off temperature. The 11.5°C increase was near the beginning of the test, while the 8.5°C increase was measured after the heaters had run over three hours and warmed up the ground somewhat.

In the next section the air velocity and volume output from the fan operating at various speeds is discussed. At all speeds tested, the air velocity immediately adjacent to the fan was many orders of magnitude above the ASHRAE[3] draftiness threshold value of 0.23 m/s. Measurement of air movement in different locations in the tent, at various distances from the fans, was not carried out. The trade-off procedure to choose the fan speed which is most effective in reducing thermal gradient but is not objectionable regarding draftiness thus became

[3]ASHRAE Handbook and Product Directory, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, 1973.

a qualitative decision. None of the three fan speeds tested seemed to cause an objectionable, excessively drafty condition. The fan speed of 700 RPM seems adequate in terms of gradient reduction; the slight improvement at 800 RPM does not seem warranted when one considers the extra power requirement and the added noise. The gradient with the fans at 700 RPM, a 3.4°C/m, is below the goal of 3.6°C/m.

FAN VOLUME OUTPUT AND DC MOTOR POWER DRAW

The volume and velocity of the air blown by the fan was determined experimentally. A J–TEC model VA–220 airspeed sensor was attached to a steel rod over the fan, which was traversed across the fan 5 cm above the blades. A velocity reading was taken every 1.3 cm. The test setup is shown in Figure 3.

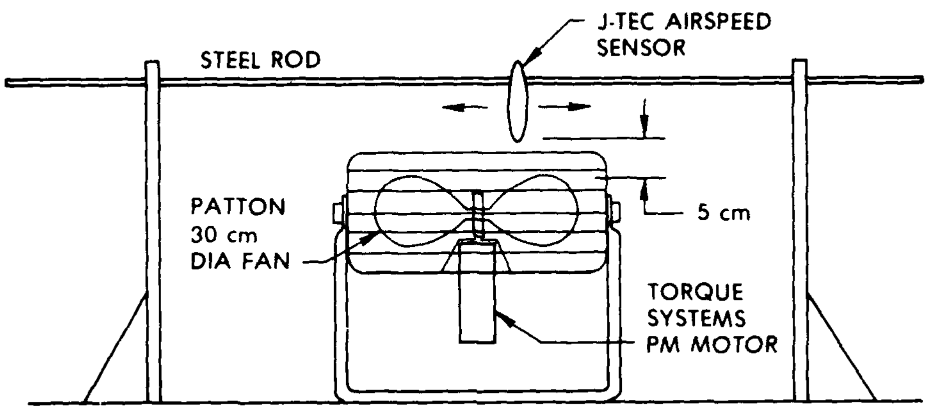


Figure 3. Test Setup for Determination of Fan Volume and Air Velocity

The volume flow of each 1.3-cm-wide annulus was found by multiplying airspeed times annulus area; all annular flows were summed to find the volume output of the fan. Table 3 gives peak air velocity and volume data for four fan speeds.

Table 3. Air Volume and Peak Velocity for Patton 3-Blade, 30-cm-Diameter Fan

Fan Speed (RPM)	Peak Velocity (m/s)	Air Volume (m^3/min)
500	2.6	8.2
600	3.0	9.4
700	3.6	10.9
800	4.1	12.6

The power required to operate the DC motor was much less than expected. Table 4 indicates the electrical requirements for the four speeds.

Table 4. DC Mot . Electrical Requirements at Four Fan Speeds

Fan Speed (RPM)	Motor Electrical Data Torque Systems MH2130 Volts	Milliamps	Watts
500	5.10	390	2.0
600	6.23	500	3.1
700	7.47	660	4.9
800	8.62	850	7.3

The power to rotate the fan at 700 RPM was less than 5 watts. This low power requirement suggested the use of thermoelectrics.

THERMOELECTRICS

Thermocouples were first considered as a means of thermoelectric power. The type E thermocouple, chromel-constantan, was chosen since it has the highest output of any standard metallic thermocouple. A wire diameter of 0.4 mm (26 gage) was chosen as a reasonable compromise between wire size and resistance. It was assumed a temperature difference of 400°C could be reached at maximum heater output. Assuming a wire length of 15 cm, output per thermocouple would be 19 mA at 29 mV or 0.55 mW. To get 5-W output, 9100 thermocouples would be needed. The difficulties of making such a thermopile, the unwieldiness, and the weight and expense involved, led us to consider semiconductor thermoelectric devices.

The thermoelectric (TE) modules used to cool small electronic devices on circuit boards were investigated next. These devices are used in the opposite sense that we are interested in, that is, DC current is supplied and a temperature difference, or a hot and cold junction, are created. The maximum temperature most standard modules can withstand is 150°C, with high temperature modules limited to 200°C. The temperature limitations are due to thermal stresses and the melting temperature of the solder used in fabrication of the devices. A special extra high temperature (275°C) TE module was under development at Borg Warner during this study. This unit may be feasible for our application, but was unavailable for evaluation.

Teledyne Energy Systems, Timonium, MD, manufactures thermoelectric generators with outputs up to 90 W for use in remote locations where electricity is unavailable. The unit, called TELAN, is powered by propane, butane or natural gas and is designed to run continuously and unattended. The heart of the TELAN unit is a 5-W thermoelectric module made from bismuth telluride semiconductor material. The module has 83 semiconductor couples in series; each couple develops 2.1 A at 29 mV from a ΔT of 222°C. The current path is a horizontal "S" shape, with adjacent couples electrically isolated by insulating material. The module is illustrated in Figure 4.

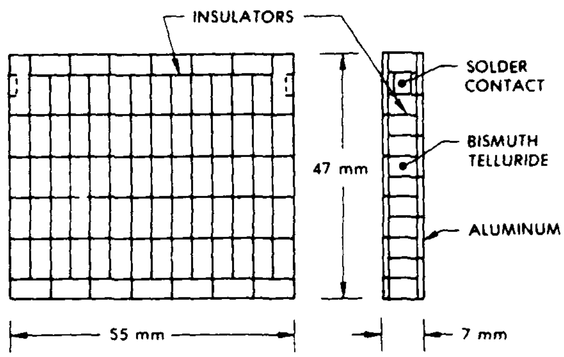


Figure 4. Thermoelectric Module

When one face of the module is held at 288°C and the other face is at 66°C (ΔT of 222°C), the module will output 2.1 A at 2.4 V, or 5.0 W when the load resistance matches the 0.9-ohm module internal resistance. When the load resistance is greater than 0.9 ohm, a higher voltage and lower current is developed, but the total power generated is lower. For example, for a load resistance of 2 ohms, the module will develop 1.2 A at 3.4 V, a power output of only 4.1 W. The module has a maximum temperature limit of 316°C, beyond which the semiconductor material degrades and the output solder connections melt. The TE module can operate continuously in a short-circuited or an open-circuit condition. The cost of a small quantity of these modules in 1981 was $78 each. If a large quantity were ordered, and if more automation were used in their manufacture, it is estimated the unit price would be 50 to 70% lower.

Global Thermoelectrics, Bassano, Alberta, Canada also manufactures thermoelectric generators. Global's unit is similar in application to Teledyne's, designed for use as a remote unattended power generator. The primary difference is this unit features a lead telluride semiconductor material. This type of material can withstand a hot junction temperature of almost 600°C. The semiconductor material is of a special geometry for installation in a cylindrical TE converter unit. The individual lead telluride components are not available commercially. The Electronics Research and Development Command (ERADCOM) in Ft. Monmouth, NJ has developed a thermoelectric generator for military field applications which uses Global's TE converter unit. The ERADCOM unit develops 625 W DC or 500 W AC. The heat source is a liquid hydrocarbon burner that features an ultrasonic atomizer. The hot junction temperature is 561°C and the cold junction temperature is 162°C. There are 256 couples in series; each one develops 22.7 A at 110 mV with a ΔT of 400°C. At this temperature difference, only two couples are needed to get 5 W, with electrical output 22.7 A at 0.22 V. A DC to DC converter would be necessary to increase the voltage to a high enough value to power a motor.

Two Teledyne Energy System 5-W TE Modules were purchased. The next task was to design a device in which to mount the module, with appropriately sized hot and cold side heat exchangers to obtain the desired hot and cold junction temperatures.

DESIGN OF HEAT EXCHANGERS

To increase the heat transfer area on both the hot and cold side of the TE module, a group of parallel vertical fins was chosen. To analyze heat flow, a single fin of rectangular profile was considered, as shown in Figure 5.

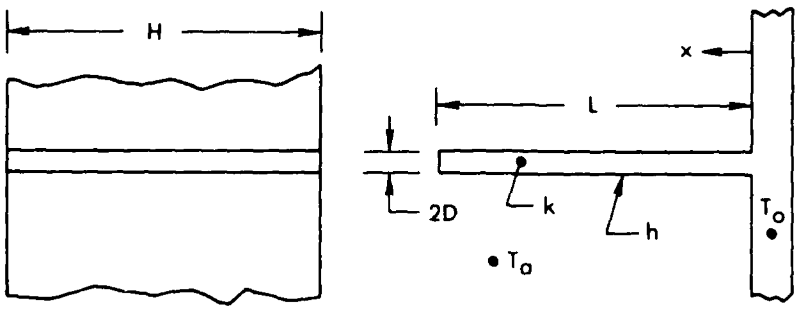


Figure 5. Fin Geometry

From Schneider[4], the heat dissipated by the fin is given by

$$q = 2H\,(T_o - T_a)\sqrt{khD}\,\tanh\left[L_c\sqrt{h/kD}\right] \qquad (1)$$

where q = heat dissipated, W

H = fin height, m

T_o = wall temperature, °C

T_a = air temperature, °C

k = thermal conductivity of fin, W/°Cm

h = convective heat transfer coefficient, W/°Cm²

D = fin half-width, m

L_c = L + D, m

[4] P. J. Schneider, Conduction Heat Transfer, Addison-Wesley, 1955.

This equation assumes heat flow to occur in the x direction only. Since our fins will have $L \cong H$, some error will be present. This analysis is used only as a guide to fin design, so the error due to the assumption of *one-dimensional* heat flow is not important. This equation is a simplified solution for a straight rectangular fin, in which fin end effects are accounted for by considering a fin of extended length $L_c = L + D$.

Design of the inner and outer fins is now discussed. *Figure* 6 shows the inner fins, the thermoelectric module, and the outer fins. The fin design is based on the requirements of the TE module: the maximum hot side temperature, T_h, was chosen as 300°C (slightly below the maximum of 316°C), and a temperature difference of 200°C is desirable for maximum power output. This gives a value of T_c of 100°C.

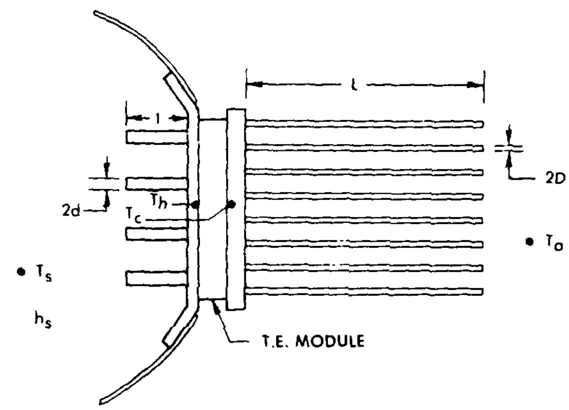


Figure 6. Heat Exchanger Unit

Previous tests with military field space heaters have indicated that the stack gas temperature, T_s, at maximum fire is about 750°C. A reasonable estimate for the outer fin ambient temperature, T_a, is 20°C.

It was necessary to estimate values of the convective heat transfer coefficient for the inner fins, h_s, and the outer fins, h_a. First the outer fins are considered. For a flat plate in laminar flow, Kreith states the average convective heat transfer coefficient to be[5]

$$h_a = 0.664 \frac{k_f}{H} Re^{\frac{1}{2}} Pr^{\frac{1}{3}} \qquad (2)$$

[5] F. Kreith, *Principles of Heat Transfer*, International Textbook Co., 1967.

where k_f = thermal conductivity of air, W/°Cm

H = plate length (in our case fin height), m

Re = Reynolds number at distance H from front of plate = VD/ν

ν = kinematic viscosity of air at 20°C, m^2/s

v = air velocity, m/s

Pr = Prandtl number

To calculate Re, it was assumed the fan was turning at 700 RPM and was blowing air at 20°C over the fins at v = 3.6 m/s (from Table 3). A fin height of 76 mm was chosen as a reasonable starting value. From Kreith, $\nu = 1.366 \times 10^{-4}$ m^2/s which gave Re = 18071. This value is considerably below the laminar to turbulent transition value of Re of 5×10^5, so the flow is definitely laminar. From tables in Kreith, for air at 20°C, Pr = 0.72 and k_f = 0.025 W/°Cm. The value of h_a was then calculated from equation (2) to be 26.3 W/°Cm².

For the inner fins, a similar procedure was followed. For air at 750°C, tables in Kreith give k_f = 0.066 W/°Cm, Pr = 0.733, and $\nu = 1.24 \times 10^{-4}$ m^2/s. A value of the speed of the ascending stack gases for a heater burning 37 cc/min diesel fuel with 63% excess air (9% CO_2) was calculated[6] to be 3.4 m/s. Assuming the fin height to be 50 mm, the Reynolds number was 1367. From equation (2) the inner fin convective heat transfer coefficient was h_s = 29.2 W/°Cm².

The heat transfer through the TE module with a temperature differential of 200°C was next needed. From a handbook[7], the thermal conductivity of bismuth telluride, Bi_2Te_3 is 1.98 W/°Cm. The module dimensions minus the between-couple insulating strips are 51 by 44 mm, with the bismuth telluride portion of the thickness 5 mm. There is an aluminum cap on either side of the bismuth telluride which will be ignored in these calculations since the aluminum conducts heat about 100 times better than bismuth telluride.

The equation for heat conduction, q, is

$$q = \frac{Ak(\Delta T)}{B} \qquad (3)$$

[6]W. Nykvist, unpublished notes.

[7]Comprehensive Inorganic Chemistry, Vo. 2, Permagon Press, 1973.

where A = heat transfer area, m^2

k = thermal conductivity, W/°Cm

ΔT = temperature difference, °C

B = conduction thickness, m

For the above conditions, q = 177.7 W

Given this value for heat flow, the inner and outer fins were sized using equation (1). For the outer fins, solving equation (1) for L gave

$$L = \frac{\tanh^{-1}\left[\dfrac{q}{2FH(T_c - T_a)\sqrt{h_a kD}}\right]}{\sqrt{h_a/kD}} - D \qquad (4)$$

where F = number of fins

k = thermal conductivity of aluminum, 206 W/°Cm

Aluminum was chosen for the fin material due to its high thermal conductivity. As the number of fins increases, L decreases. An estimate of F of 8 was used as a reasonable value. The value assumed for 2D was 2.3 mm (0.090 in.), a reasonably thick aluminum standard sheet size on hand in our machine shop. Using the value of h_a previously calculated, 26.3 W/°Cm, a fin height of 76 mm, T_c = 100°C, and T_a = 20°C, equation (4) gave L = 87 mm.

For the inner fins a similar procedure was followed to determine fin thickness, 2 l. The same value of q (177.7 W) was used. In equation (4),$(T_c - T_a)$ was replaced with $(T_s - T_h)$; values used were T_s = 750°C and T_h = 300°C. A fewer number of thicker fins was chosen; 2d = 3.2 mm (0.125 in.) and F = 4 was assumed. The fin height was assumed to be 50 mm, and the previously calculated value of h_s = 29.2 W/°Cm was used. Solving (4) with these values gave l = 33 mm.

HEAT EXCHANGER FABRICATION AND TEST

The previous section gave dimensions for the size of the inner and outer finned heat exchangers. That analysis, far from exact, provided a good starting point in fabricating a practical device. The recommended design featured an outer heat exchanger with eight fins 87 x 76 x 2.3 mm and an inner heat exchanger with four fins 33 x 50 x 3.2 mm.

The heat exchangers were mounted on the inside and outside of a 102-mm (4-in.) diameter exhaust stack from a field space heater. A rectangular hole was cut in the stack, the inner heat exchanger was attached to the stack, and the outer heat exchanger, sandwiching the TE module, was attached to the inner heat exchanger. The outer heat exchanger then is cooled by the downward airflow from the fan, as shown in Figure 7.

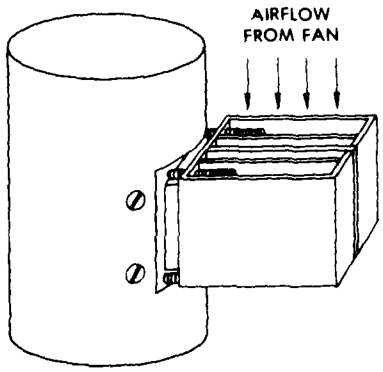


Figure 7. Mounting of Heat Exchanger Unit

The outer heat exchanger was fabricated from aluminum, using two different fin thicknesses. The outside fins were 3.2 mm thick, made from two 22 x 76 x 3.2-mm aluminum U channels welded face to face with a 76 x 76 x 2.3-mm plate between. Prior to welding the U channels together, two 2.3-mm fins were welded inside the U with 6.3-mm spacing. The size of the available aluminum channel limited the fin length to 76 mm, and practical considerations limited the number of fins to seven. The end surface, however, can be regarded as the eighth fin. Total fin surface area is approximately 85% of the value recommended in the analysis. Design 1 of the heat exchanger unit is shown in Figure 8.

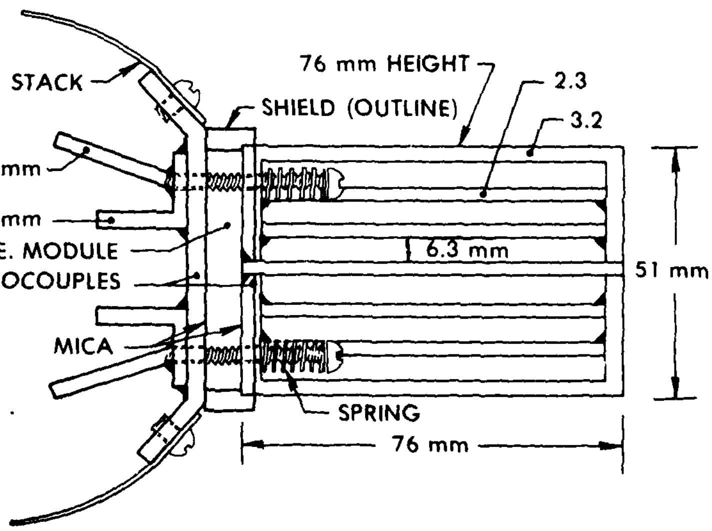


Figure 8. Heat Exchanger Unit, Design 1, Top View

The inner heat exchanger was fabricated from two aluminum angles 19 x 19 x 3.2 mm, 80 mm long, and two angled aluminum plates, 25 x 50 x 3.2 mm. The total inner fin surface area is approximately 90% of the recommended value.

The two heat exchangers were held together by spring force. Four 8–32 machine screws with compression springs pass through holes in the outer heat exchanger and are screwed into tapped holes in the inner heat exchanger. The spring force is adjusted by tightening or loosening the machines screws. The springs are necessary to keep the heat exchangers in good and even thermal contact with the TE module. The small size of the machine screw and the lack of good thermal contact it has with the outer heat exchanger keep heat transfer through the screw to a minimum. The fins below the machine screw were reduced in height to provide clearance for the screw. Both surfaces in contact with the TE module were ground flat.

The TE module must be electrically insulated from the two surfaces it contacts, but must have good thermal contact. Mica was used as an electrical insulator, and a non-electrically conductive thermal heat-conducting paste was applied to both sides of each thin piece of mica.

The test setup to evaluate this Design 1 unit was as shown in Figure 9.

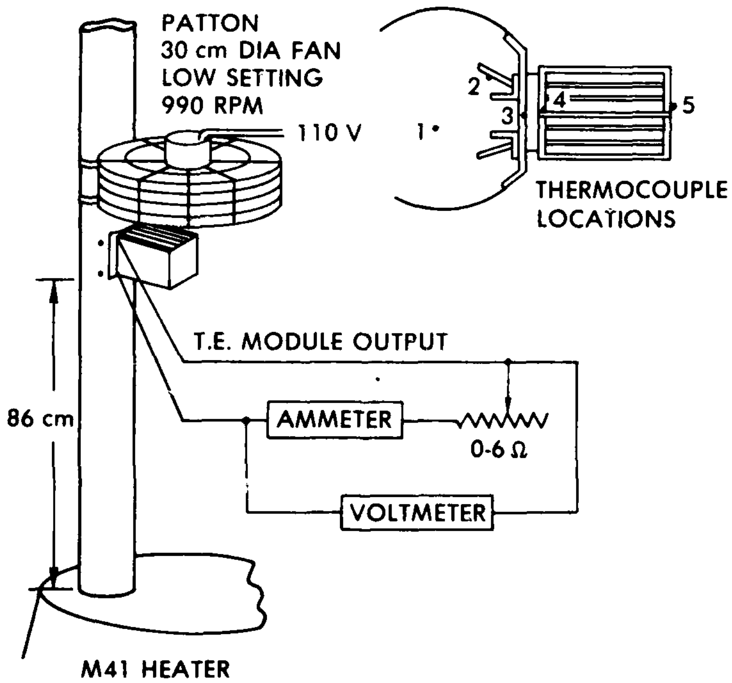


Figure 9. Test Setup for Heat Exchanger Unit

The variable resistive load, 0 to 6 ohm, permitted a matching of the internal resistance of the TE module so maximum output could be achieved. The M41 heater was fired with gasoline and was in its standard configuration except for the modified stack section. An oversight in this test was using the fan with AC motor at the low setting where it developed 990 RPM. A variable transformer should have been used to lower the speed to a more realistic 500 to 700 RPM. Test results are shown in Table 5.

Table 5. Performance of Heat Exchanger Unit, Design 1

Fuel	Temperatures (°C)						Load	TE Output		
cc/min	1	2	3	4	5	ΔT	Ohms	Volts	Amps	Watts
16.4	371	165	149	54	43	95	0.2	0.5	2.1	1.1
							0.7	1.0	1.5	1.5
							0.9	1.5	0.8	1.2
							4.6	1.8	0.4	0.7
22.0	465	215	193	63	43	130	0.2	0.5	2.9	1.4
							0.4	1.0	2.3	2.3
							0.9	1.5	1.7	2.6
							1.7	2.0	1.2	2.4
							4.0	2.5	0.6	1.6
41.4	654	354	316	104	71	212	0.3	1.0	3.4	3.4
							0.5	1.5	2.9	4.4
							0.8	2.0	2.6	5.2
							1.2	2.5	2.1	5.25
							1.9	3.0	1.6	4.8
							3.0	3.6	1.1	4.0

At high heater output, the temperatures for T_h and T_c of 316 and 104°C are remarkably close to the values of 300 and 100°C used in the analysis to design the heat exchangers. The maximum output of 5.25 W with a ΔT of 212°C is slightly higher than expected; the standard module achieves 5.0 W with a ΔT of 222°C. The difference is probably due to our higher hot junction temperature, 316 v.s. 288°C for the standard module test point. Also, the cold junction temperature, recorded near the center of the module, may not be representative of the overall module surface. An average cold junction temperature lower than the 104°C recorded would result in a higher output.

As expected, peak power was developed when the load resistance was in the range of the TE module internal resistance, 0.7 to 0.9 ohm. The load resistor, an Ohmite rheostat, 25-W, 0 to 6 ohm, was adjusted to provide a 0.5-V increase in the output voltage. The value of resistance for a given rheostat setting was computed using Ohm's law. Thick wires (16-gage), as short as possible, were used to connect the TE module to the load, to keep resistive losses in the wire to a minimum. The hot junction temperature of 316°C was exactly the maximum recommended value. This leaves no room for error; 316°C could be exceeded with a fuel metering valve calibrated slightly higher. Rather than downsize the inner fins, a decision was made to enlarge the outer fins. In the test work, it was noticed that only the outer half of the outer fins were in the strong airflow from the fan, due to the protective cage surrounding the fan standing the fan blade off from the stack 2 to 3 cm, and the slower airspeed near the outer tip of the blades. Also, the fan speed used in the test was too high, 990 RPM, and lowering it to 500–700 RPM would raise T_c and T_h.

The outer heat exchanger was redesigned. Figure 10 shows Design 2 of the heat exchanger unit.

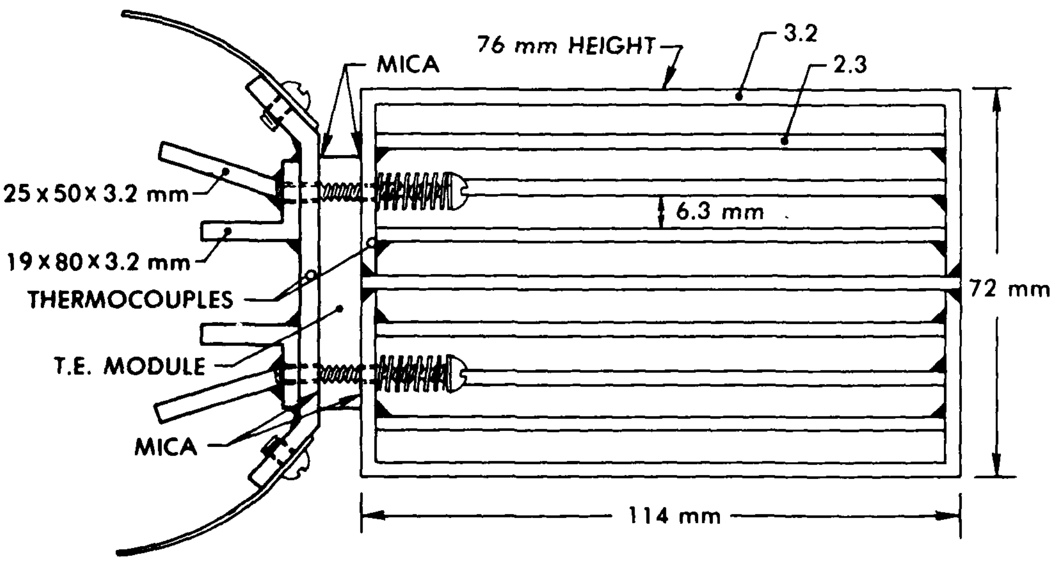


Figure 10. Heat Exchanger Unit, Design 2

The fin height remained the same as in Design 1, but the number of fins was increased by two and the length was increased by 50% to 114 mm. This larger outer heat exchanger was fabricated in the same manner as was Design 1, except a 114 x 35 x 3.2-mm aluminum channel was used. The total outer fin surface area for Design 2 is 165% of the recommended area. Tests using this heat exchanger also used a small DC fan motor, driven by thermoelectric power. Results are given in a following section, Prototype Test Results, in Table 7.

MOTOR SELECTION

Obtaining a motor to drive the fan proved to be a difficult problem. To start, it was necessary to estimate the torque required to rotate the fan at 700 RPM. The torque value was estimated by examining the specifications of the Torque Systems MH 2130 DC motor used in the original tent tests. Rotating the fan at 700 RPM, the Torque Systems motor drew 660 mA at 7.5 V. Specifications for this motor are:

torque sensitivity	7.9 N-cm/amp
static friction	2.1 N-cm
viscous friction	0.35 N-cm/1000 RPM

At 700 RPM, the sum of static and viscous friction is 2.34 N-cm. The torque sensitivity times the current less the friction is the developed torque, which was calculated to be 2.9 N-cm. The work W developed by the motor, in watts, is calculated by

$$W = (\text{torque in N-cm})\ (\text{RPM})\ (1.05 \times 10^{-3})$$

The work generated by the Torque Systems motor was 2.13 W. Input power was 4.95 W, so efficiency was 43%. This low efficiency is due to the very light load on a motor rated at 93 W continuous output.

Our requirements for a thermoelectrically powered DC permanent magnet motor can now be stated:

input power	2.1 A @ 2.5 V
torque	2.9 N-cm
speed	700 RPM
internal resistance	0.8–1.0 ohm

The low voltage requirement eliminates dozens of motor manufacturers from consideration, leaving only a few to contact. No manufacturer that I contacted had a motor that met these requirements, and none were interested in specially making a motor without the potential of an order for a substantial quantity. Buehler Products, Inc. Kinston, NC did send a motor they thought to be appropriate for our application, but our tests indicated it was undersized, drawing 5.3 A at 3.6 V (18.8 W) to rotate the fan at 700 RPM.

Since no direct-drive motor could be obtained, a search was made to get a gear-reduced motor to meet our requirements. I had avoided considering a gear reducer since it involves more frictional losses than direct drive, and with only 5 W of available power, gear reducer friction becomes significant. A Marx motor, model 6301, was purchased from Aristo-Craft, NY, NY. This 6 V motor comes with a universal planetary gear reducing unit, which includes 3:1, 4:1, 5:1 and 6:1 reductor units. The motor is 40 mm in diameter, 74 mm long less gear reducer, and is rated at 20 W when operated at 6 V. From the speed-torque data in the Marx catalog, this motor, at 2.5 V and 2.1 A, develops 3500 RPM at 0.9 N-cm. Using a 4:1 gear reducer with an estimated 80% efficiency would result in an output of 2.9 N-cm at 700 RPM. This is exactly our requirement. The fan was attached to the Marx motor and the unit was tested using a DC power supply and strobe light. Three gear reducer ratios were tested, 3:1, 4:1 and 5:1 with results shown in Table 6.

Table 6. Marx Motor Performance with Three Gear Reducers

Gear Reduction	Fan Speed (RPM)	Electrical Requirements Volts	Amps	Watts
3:1	600	1.9	3.1	5.9
	700	2.3	3.9	9.0
	800	2.6	4.7	12.2
4:1	600	2.2	2.7	5.9
	700	2.6	3.4	8.8
	800	2.9	4.0	11.6
5:1	600	2.7	2.2	5.9
	700	3.1	2.6	8.1
	800	3.6	3.2	11.5

The power needed to operate the fan was more than expected, with 5.9 W necessary to achieve only 600 RPM. The higher tnan expected power draw may be due to a higher actual torque value than previously calculated, or high bearing and gear reducer friction losses.

The steel fan blades and hub put a substantial radial load on the sintered metal sleeve bearings, increasing frictional losses. The plastic panetary gears and bearings are also subject to a substantial radial load, increasing frictional losses as well.

The internal resistance of the motor was measured to be 3.8 ohms using the Data Precision digital multimeter. This is substantially more than the 1 ohm desired, and will mean the TE module will operate with reduced power, but with higher voltage and lower amperage than the 2.5-V, 2.1-A, matched-resistance values. With this in mind, the 5:1 gear reducer was attached to the motor for testing with the TE module output.

PROTOTYPE TEST RESULTS

The test setup to evaluate the heat-driven fan prototype is similar to that shown in Figure 9 except the rheostat load is replaced with the Marx motor with 5:1 gear reduction. A Strobotac 631 strobe was used to measure fan speed. The fan is the same one used throughout this test program, a 3-bladed, 30-cm-diameter Patton fan in protective cage. The fan cage was clamped to the stack with radiator-type clamps. The Design 2 outer heat exchanger was used. For this test, diesel fuel was burned in the M41 heater. Results are given in Table 7.

Table 7. Performance of Heat Driven Fan Prototype

Fuel	Fan	TE Output			Temperatures (°C)						
cc/min	RPM	Volts	Amps	Watts	1	2	3	4	5	AMB	ΔT
14.1	288	1.6	0.9	1.5	370	190	180	77	54	25	103
20.0	416	2.2	1.2	2.8	488	250	227	82	54	28	145
27.6	507	2.5	1.5	3.8	570	300	274	93	60	27	181
37.5	566	2.7	1.7	4.6	632	346	318	104	63	32	214

A temperature of 318°C was reached on the TE module hot side even though the larger outer heat exchanger was used. Although this exceeds the maximum module temperature of 316°C, it is pointed out that this test was run in August, and temperatures in the 3 x 3.5 x 2 m test shelter reached 32°C (89°F). More realistic test conditions may confirm the heat exchangers are sized properly; if T_h is still too high, the inner heat exchanger could be reduced in size.

The output power only reached 4.8 W due to the resistance mismatch between the load (3.8 ohms) and the TE module (0.9 ohm). A better match would have resulted in the full 5.25 W being used. The maximum diesel fuel flow rate was limited to 37.5 cc/min. This is the maximum diesel fuel flow rate achieved in previous NLABS tests of the M41 heater in cold weather.[8]

[8]W. Nykvist, Evaluation of Liquid Fuel Space Heaters: Standard Military Developmental, and Foreign, US Army Natick Research and Development Laboratories, Natick, MA, Technical Report NATICK/TR–79/021, October 1978 (AD A075800).

DISCUSSION

Unavailability of an appropriate motor was the primary reason for the limited success of the heat-driven fan prototype. The fan speed of 566 RPM that was achieved, however, was over 80% of the design goal of 700 RPM.

A higher fan speed could be achieved if several areas of the design were improved, such as:

(a) improved motor characteristics, with

- internal resistance approximately 1 ohm
- direct drive, eliminating gear reducer
- ball bearings
- low resistance brushes

(b) use of lighter-weight hub and fan unit

(c) fan blade design optimized

If the above requirements are not enough to achieve satisfactory performance, use of a larger TE module or two 5-W modules would be considered. The manufacturer of the TE modules could be requested to fabricate a 7-W module, for instance. If the low voltage is too problematic, a DC to DC converter could be employed to increase voltage. These devices operate with about 75% efficiency, and cost $40 to $80 each. One must be careful not to make the heat-driven fan too expensive, however. A non-essential item such as a heat-driven fan would probably be purchased by the users from Army stock funds. The cost must be reasonable, as military commanders have only a limited amount of funds to equip their troops. A concerted effort, therefore, was made in this study to keep the cost of the unit as low as possible. A single 5-W TE module does provide enough power; optimum use of this 5 W is difficult due to the low voltage involved. A motor wound to meet our speed and load requirements – with internal resistance 0.9 ohm, low friction losses, and operating at a reasonable 65% efficiency – should achieve more than 750 RPM with a 5-W input.

One area of concern that must be dealt with in a more refined heat-driven fan design is the possibility of exceeding the maximum temperature limit of the TE module. Overfiring of the heater, or placing the heat-driven fan unit below the first stack section rather than between the second and third, or stoppage of the fan with the heater at high fire could all result in thermal damage to the TE module. A design that accommodates these conditions may result in a low power output at normal operating conditions. A higher capacity TE module that could generate 5 W at a 250°C hot junction temperature, for example, would allow a 66°C cushion at the hot junction. A higher temperature semiconductor material, such as bismuth antimony (370°C), may be a solution.

A literature search for propeller fan design criteria was made with negative results. At these low speeds, there may be a much more efficient blade design to move the desired quantity of air. The number of blades, pitch angle, and the blade diameter are variables which could be optimized. An intensive investigation of fan blade design could result in a design with more airflow using less power, yielding a more effective heat driven fan. A lighter weight hub and fan blade unit would reduce friction and improve fan speed for any blade design.

The maximum fan speed of 700 RPM is achieved only at a high heater output. At lower heater settings, a lower and less effective fan speed is achieved. At 600 RPM (see Table 2), the thermal gradient was reduced 51% as opposed to a 62% reduction for 700 RPM. No gradient tests were run at 500 RPM, but it is estimated the gradient reduction would be 40–45%. Without further tests with an appropriate DC fan motor, it is difficult to correlate fan speed with heater firing rate. In cold weather the heater is seldom used below half capacity, approximately 20 cc/min. From Table 7, the fan speed at half capacity, 416 RPM, is 73% of the high capacity value of 566 RPM. If this ratio holds true for a maximum fan speed of 750 RPM, the half-capacity fan speed would be 550 RPM. Over the effective operating range, therefore, a cold weather thermal gradient reduction could be expected to be between 50 and 65%.

Since tests of the prototype unit could not be held up until cold weather set in, they had to be conducted in August. The flow rate of the diesel fuel was set at an expected maximum cold weather value of 37.5 cc/min. The specification for the metering valve,[9] however, permits a somewhat wide calibration range of 38 to 44 cc/min with gasoline at 10 to 21°C at setting 7. Under not very cold weather conditions with a valve calibrated on the high side, at the maximum knob setting of 9, flow rates of well over 50 cc/min are possible with gasoline, and well over 40 cc/min are possible with diesel fuel. If any further development of a heat-driven fan is undertaken, a study of maximum firing rates under a variety of conditions should be made. From such a study a realistic value of the maximum stack gas temperature needed to design the inner stack heat exchanger could be determined.

CONCLUSIONS

An investigation of the feasibility of using a thermoelectrically powered air-circulating fan to reduce the thermal gradient in Army tents is documented. Specific conclusions are:

(1) The vertical thermal gradient in heated Army tents is typically 8 to 10°C/m.

(2) The air circulation from a fan mounted above the heater, blowing downward, can dramatically reduce the gradient throughout the tent. A 30-cm fan rotating at only 700 RPM will reduce the gradient 62%.

(3) The power required to rotate the fan at 700 RPM is approximately 5 W.

[9] Military Specification, MIL–C–43343A, Control, Fuel Flow, Oil Burner, 13 Dec 74.

(4) Exhaust stack heat can be used to energize a thermoelectric module to power the fan motor. Using a commercially available TE module sandwiched between inner stack and outer heat exchangers, a temperature difference of 200°C will generate 5 W.

(5) An appropriate off-the-shelf motor to power the fan from the low voltage generated could not be located. A rough heat-driven fan prototype with a poorly matched motor did achieve limited success, 566 RPM at high heater fire, 80% of the design goal of 700 RPM.

(6) The feasibility of using available exhaust stack heat to thermoelectrically power a fan to substantially reduce the thermal gradient in tents has been demonstrated. Possible design problem areas were discussed, and design improvements were outlined to make the prototype unit fully successful.

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9. Military Specification, MIL–C–43343, Control, Fuel Flow, Oil Burner, 13 Dec 74.

APPENDIX

TEST DATA

Table A–1. Heat Stratification Test, 4.9 x 4.9 m Frame Type, 28 Nov 80

Time @ (min)	Fan Speed (RPM)	Ambient Temp (°C)	Temperature (°C) At Height Above Ground of: 0.3 m	1.2 m	1.8 m	2.5 m	Thermal Gradient (°C/m) 0.3–1.8 m	0.3–2.5 m
Start		3.5	All Temps at 3.5°C					
60 @	0	4.0	13	18	21.5	25.5	5.7	5.7
30 @	600	5.2	18.3	20.5	22.3	21.0	2.7	1.2
30 @	700	5.5	19.5	22	23	21.5	2.3	0.9
30 @	800	5.7	19.5	22	23	21	2.3	0.7
30 @	0	5.7	16	20	24	26	5.3	4.5

Weather: Moderate Rain During Entire Test
Winds: Light
Test Period: 8:30–11:30 A.M.

Table A–2. G.P. Medium Tent Heat Stratification Test, 11 Dec 80

Time @ (Min)	Fan Speed (RPM)	Ambient Temp (°C)	Position in Tent	Temperature (°C) At Height Above Ground of: 0.3 m	1.2 m	1.8 m	2.2 m	2.9 m	Thermal Gradient (°C/m) 0.3–1.8 m	0.3–2.9 m
Start		–9.5		All Temps –9.5°C						
170 @	0	–5	N. End	11	20	24.5	27		9.0	
			Center	12	21	25.5		30.5	9.3	7.1
			S. Cor	12	20.5	25.5	28		9.0	
30 @	600	–5	N. End	14.5	21.5	23	23.5		5.7	
			Center	14.5	21.5	22.5		24	5.3	3.9
			S. Cor	16.5	21.5	23	23.5		4.3	
40 @	700	–4.5	N. End	15	19.5	20.5	21		3.6	
			Center	16.5	20.5	21.3		22	3.2	2.3
			S. Cor	16.7	19.5	20.5	21		2.5	
60 @	0	–5.5	Center	10.5	17.7	21.5		27.5	7.3	6.5

Weather: Sunny
Winds: Avg 3 m/s, Gusts to 8.8 m/s, from WNW
Test Period: 10:00 A.M.–3:00 P.M.

Table A–3. G.P. Medium Tent Heat Stratification Test, 12 Dec 80

Time @ (Min)	Fan Speed (RPM)	Ambient Temp (°C)	Position in Tent	Temperature (°C) At Height Above Ground of: 0.3 m	1.2 m	1.8 m	2.2 m	2.9 m	Thermal Gradient (°C/m) 0.3–1.8 m	0.3–2.9 m
Start		–11		All Temps –10°C						
150 @	0	–4	N. End	10	17.5	21.5	24.5		7.7	
			Center	11.5	19.5	23.5		28	8.0	7.5
			S. Cor	10	17.5	22.5	25		8.3	
20 @	600	–3.5	N. End	17	21.5	22.5	23.5		3.7	
			Center	17.5	23	23.5		24	3.7	2.9
			S. Cor	17.5	21.5	23	23		4.0	
20 @	700	–2.5	N. End	19.5	23.5	24	24.3		3.0	
			Center	21	24.9	25.2		26.2	2.8	2.1
			S. Cor	19.5	23.5	24	24.3		3.0	
30 @	800	–3	N. End	19.5	23	24	NR		3.0	
			Center	23	24.5	25		25	1.3	0.9
			S. Cor	20	22.6	23.5	23.5		2.3	
45 @	0	0	N. End	16.5	25	29.5	32.5		8.7	
			Center	17.5	27	32		37	9.7	8.2
			S. Cor	18	26.5	31.5	35		9.0	

Weather:	Clear, Sunny
Winds:	Slight, Westerly
Test Period:	8:00 A.M.–12:25 P.M.

Table A–4. G.P. Medium Tent Heat Stratification Test, 13 Jan 81

Time @ (Min)	Fan Speed (RPM)	Ambient Temp (°C)	Position in Tent	Temperature (°C) At Height Above Ground of: 0.3 m	1.2 m	1.8 m	2.2 m	2.9 m	Thermal Gradient (°C/m) 0.3–1.8 m	0.3–2.9 m
Start		–19		All Temps –16.5°C						
75 @	0	–16.5	Center	2.5	11	18.5		21	10.7	8.4
30 @	700	–15	N. End	12	16.5	17.5	18		3.7	
			Center	13	18	19		18.5	4.0	2.7
			S. Cor	11.5	16.5	17.5	18		4.0	
25 @	800	–13.5	N. End	14	18	19.5	19.5		3.7	
			Center	16	20	21		21.5	3.3	2.5
			S. Cor	13.5	18	19	19.5		3.7	
25 @	700	–12.5	N. End	15.5	20.2	21.5	22		4.0	
			Center	17.5	22.5	22.8		22.5	3.5	2.5
			S. Cor	15	20.3	21.5	21.5		4.3	
40 @	0	–12	N. End	7	15.2	21	24.5		9.3	
			Center	9	17.5	23.5		28	9.7	8.6
			S. Cor	7.5	15.2	21	25		9.0	

Weather: Cloudy
Winds: Slight
Test Period: 7:15 A.M.–10:30 A.M.
Heaters: M1941, Gasoline @ 37.5 cc/min

LIST OF SYMBOLS

A	Heat transfer area, m^2
B	Heat conduction thickness, m
D	Half-thickness of outer fin, m
d	Half-thickness of inner fin, m
F	Number of fins
H	Height of fin, m
h	Convective heat transfer coefficient, $W/^{\circ}Cm^2$
h_a	Convective heat transfer coefficient for outer fins, $W/^{\circ}Cm^2$
h_s	Convective heat transfer coefficient for inner fins, $W/^{\circ}Cm^2$
k	Thermal conductivity of fin, $W/^{\circ}Cm$
k_f	Thermal conductivity of fluid (air), $W/^{\circ}Cm$
L	Fin length, outer heat exchanger, m
L_c	Extended fin length, outer heat exchanger, m
l	Fin length, inner-stack heat exchanger, m
Pr	Prandtl number
q	Heat flow, W
Re	Reynolds number
T_a	Temperature of air inside tent, $^{\circ}C$
T_c	Temperature of cold side of TE module, $^{\circ}C$
T_h	Temperature of hot side of TE module, $^{\circ}C$
T_o	Temperature at base of fin, $^{\circ}C$
T_s	Temperature of stack gases, $^{\circ}C$
ΔT	Temperature difference, $^{\circ}C$
v	Air velocity, m/s
ν	Kinematic viscosity of air, m^2/s